Experimental Studies on Heat Transfer Augmentation Using TMT Rods with and without Baffles as Inserts for Tube Side Flow of Liquids

A thesis submitted in partial fulfilment of the requirements for the degree of

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In
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Under the Guidance
of

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CERTIFICATE

This is to certify that the thesis entitled, **"EXPERIMENTAL STUDIES ON HEAT TRANSFER AUGMENTATION USING TMT RODS WITH AND WITHOUT BAFFLES AS INSERTS FOR TUBE SIDE FLOW OF LIQUIDS"** submitted by **Jitendra Kumar Patro & Abhinav Malviya** in partial fulfilments for the requirements for the award of Bachelor of Technology Degree in Chemical Engineering at National Institute of Technology, Rourkela (Deemed University) is an authentic work carried out by them under my supervision and guidance.

To the best of my knowledge, the matter embodied in this thesis has not been submitted to any other University / Institute for the award of any Degree or Diploma.

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ABSTRACT

This project deals with the introduction of TMT rods as inserts as passive augmentation device, in the flow path of inner tube side liquid flow. The effect of turbulence on heat transfer & pressure drop was compared with the values for smooth tube. The effect of baffles was also taken into account and again a comparative study was made on the basis of varying the baffle spacing. All the results and readings were compared with the standard data from the smooth tube. Whenever it comes to enhance the heat transfer between the surfaces or in other words augmenting the heat exchanger, the pressure drop does play an important role and becomes another important factor to be considered and to be kept in mind.

Two TMT Rods (d_i = 8 mm, 10 mm) were used for the experimental purpose. In the beginning we conducted the experiment without any insert to get the value for plane heat exchanger and thereafter the experiment was repeated with TMT Rods (d_i = 8mm, 10 mm) without any baffles and with baffles with varying baffle spacing (β=10cm, 20cm, 30cm). The maximum value of performance evaluation criteria R1 was found to be around 2.46 for 10mm insert with β = 10cm and similarly the highest value for f_a/f_o was found to be around 21.

The friction factor was found to be significantly high and that has been an area of concern and which needs to be minimized.

CONTENTS

LIST OF FIGURES

LIST OF TABLES

NOMENCLATURE

A_i	Heat transfer area, m^2
A_{xa}	Cross- section area of tube with twisted tape, m^2
A_{xo}	Cross-section area of tube, m^2
C_p	Specific heat of fluid, J/Kg.K
d_i	ID of inside tube, m
d_o	OD of inside tube, m
f	Fanning friction factor, Dimensionless
f_a	Friction factor for the tube with inserts, Dimensionless
f_o	Theoretical friction factor for smooth tube, Dimensionless
g	acceleration due to gravity, m/s^2
Gz	Graetz Number, Dimensionless
h	Heat transfer coefficient, $W/m^2 °C$
h_a	Heat transfer coefficient for tube with inserts, $W/m^2 °C$
h_o	Heat transfer coefficient for smooth tube, $W/m^2 °C$
h_i(exp)	Experimental Heat transfer coefficient, $W/m^2 °C$
h_i(theo)	Theoretical Heat transfer coefficient, $W/m^2 °C$
L	heat exchanger length, m
LMTD	Log mean temperature difference, °C
m	Mass flow rate, kg/sec
Nu	Nusselt Number, Dimensionless
Pr	Prandtl number, dimensionless
Q	Heat transfer rate, W

Re	Reynolds Number, Dimensionless
R_1	Performance evaluation criteria based on constant flow rate, Dimensionless
R_3	Performance evaluation criteria based on constant pumping power, Dimensionless
U_i	Overall heat transfer coefficient based on inside surface area, $W/m^2 °C$
v	flow velocity, m/s^2

Greek letters

Δh	Height difference in manometer, m
ΔP	Pressure difference across heat exchanger, N/m^2
μ	Viscosity of the fluid, $N\ s/m^2$
μ_b	Viscosity of fluid at bulk temperature, $N\ s/m^2$
μ_w	Viscosity of fluid at wall temperature, $N\ s/m^2$
ρ	Density of the fluid, kg/m^3
β	Baffle spacing in cm.

CHAPTER 1

INTRODUCTION

INTRODUCTION:

There are various important unit operations in chemical engineering and these unit operations are also called the hearts of chemical engineering. Heat transfer is also one of them. May it be any industry steel industry, pharmaceutical, fertilizer, Agricultural product, crystallization process, power generation everywhere heat transfer finds its significant role.

Heat transfer is basically done through heat exchanger and if any how we can improve the thermal performance of these heat transfer equipment ie. Heat exchanger it will be a great boon for the industry. By increasing the thermal performance of heat exchanger we meant making the heat transfer operation more economical and efficient. In order to achieve that, we need to modify the construction of heat exchanger, using efficient metal surface for heat transfer to take place.

Several modification and new ideas to enhance the heat transfer led to many technical terms like ***heat transfer augmentation*** also tends to increase known as ***heat transfer intensification*** or ***enhancement***. Application of augmentation technique the heat transfer coefficient but at the same time pressure drop also increases significantly. So, while applying any augmentation technique on heat exchanger analysis of both, heat transfer rate and pressure drop has to be done. Moreover long durability and economic feasibility are two other major issues that need to be addressed. To get high heat transfer rate keeping pressure drop under limit (keeping pumping cost under control), many techniques have been applied in recent years and are discussed in the following sections.

Introduction of insertions in the flow path of inner tube side liquid has been quite effective in past studies. For experimental work, TMT rods of diameter 8 mm and 10 mm are used. Effect of TMT rods with baffles of varying baffle spacing (β= 10cm, 20cm, 30cm) have been studied.

CHAPTER 2

LITERATURE REVIEW

2.1 CLASIFICATION OF ENHANCEMENT TECHNIQUES: [1, 2]

Basically all augmentation technique can be divided into three categories :

1. Passive Techniques
2. Active Techniques
3. Compound Techniques.

1. **PASSIVE TECHNIQUES:** These technique deals with the surface and geometrical modification by the introduction of inserts or any other external device in the flow path of inner tube side fluid. They give high heat transfer coefficient by disturbing the existing flow pattern (except for extended surfaces) that increases the pressure drop as well. In case of extended surfaces, effective heat transfer area of the extended surface side is increased. Passive techniques are preferred over active technique as they do not require any direct input of external power. Heat transfer augmentation by these techniques can be achieved by using:

   - **Treated Surfaces**: this method is basically used for boiling and condensing duties by using pits, cavities or scratches like alteration in the surfaces of the heat transfer area which may be continuous or discontinuous.
   - **Rough surfaces**: This is another way of disturbing viscous sub-layer region. These techniques are abundantly used in single phase turbulent flows.
   - **Extended surfaces**: finned surfaces are very much in use because they not only disturb the flow pattern but also increases the heat transfer area significantly.
   - **Displaced enhancement devices**: These inserts generally find their use in confined forced convection. They indirectly intensify heat transfer rate at the heat exchange surface by displacing the fluid from the heated or cooled surface of the duct with bulk fluid from the core flow.
   - **Coiled tubes**: This give more compact heat exchangers as they give high heat transfer coefficient in single flow by generating secondary flow and vortices due to curvature of the

coils.

2. **ACTIVE TECHNIQUES:** From the use and design point of view these techniques are more complex as it requires some external power input to maintain the desired flow modification and enhancement in the heat transfer rate. That is the reason why it is not used widely and also in comparison to the passive techniques, these techniques doesn't sound promising as in many cases it is extremely or almost next to impossible to provide an external power source. Various active techniques are as follows:

❖ **Mechanical Aids**: It includes rotating tube exchangers and scrapped surface heat and mass exchangers. These devices stir the fluid either by mechanical means or rotating the surface.

❖ **Surface vibration**: It is generally used in single phase flows. A low or high frequency is applied to vibrate the surface as a result of that we get higher convective heat transfer coefficients.

❖ **Fluid vibration**: Instead of vibrating the surface the same can be achieved by creating pulsations in the fluid itself. This kind of vibrational enhancement technique is employed for single phase flows.

❖ **Injection**: This technique is used for single phase heat transfer process. In this method, same or different fluid is injected into the main bulk fluid by a porous heat transfer interface or upstream of the heat transfer section.

3. COMPOUND TECHNIQUES: This technique is a combined form of more than one above mentioned technique and basically used with a purpose to get the higher performance from heat exchanger.

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2.2 PERFORMANCE EVALUATION CRITERIA: [1]

In most practical applications of enhancement techniques, the following performance objectives, along with a set of operating constraints and conditions, are usually considered for optimizing the use of a heat exchanger:

1. Increase the heat duty of an existing heat exchanger without altering the pumping power (or pressure drop) or flow rate requirements.
2. Reduce the approach temperature difference between the two heat-exchanging fluid streams for a specified heat load and size of exchanger.
3. Reduce the size or heat transfer surface area requirements for a specified heat duty and pressure drop or pumping power.
4. Reduce the process stream's pumping power requirements for a given heat load and exchanger surface area.

It may be noted that objective 1 accounts for increase in heat transfer rate, objective 2 and 4 yield savings in operating (or energy) costs, and objective 3 leads to material savings and reduced capital costs.

Different Criteria used for evaluating the performance of a single phase flow are:

Fixed Geometry (FG) Criteria: The area of flow cross-section (N and d_i) and tube length L are kept constant. This criterion is typically applicable for retrofitting the smooth tubes of an existing exchanger with enhanced tubes, thereby maintaining the same basic geometry and size (N, d_i, L). The objectives then could be to increase the heat load Q for the same approach temperature ΔT_i and mass flow rate *m* or pumping power P; or decrease ΔT_i or P for fixed Q and m or P; or reduce P for fixed Q.

Fixed Number (FN) Criteria - The flow cross sectional area (N and d_i) is kept constant, and the heat exchanger length is allowed to vary. Here the objectives are to seek a reduction in either the heat transfer area (A → L) or the pumping power P for a fixed heat load.

Variable Geometry (VN) Criteria - The flow frontal area (N and L) is kept constant, but their diameter can change. A heat exchanger is often sized to meet a specified heat duty Q for a fixed process fluid flow rate *m*. Because the tube side velocity reduces in such cases so as to accommodate the higher friction losses in the enhanced surface tubes, it becomes necessary to increase the flow area to maintain constant *m*. This is usually accomplished by using a greater number of parallel flow circuits.

Case	Geometry	M	P	Q	ΔTi	Objective
FG-1a	N, L, D_i	X			X	Q↑
FG-1b	N, L, D_i	X		X		ΔTi↓
FG-2a	N, L, D_i		X		X	Q↑
FG-2b	N, L, D_i		X	X		Δ Ti↓
FG-3	N, L, D_i			X	X	P↓
FN-1	N, D_i		X	X	X	L↓
FN-2	N, D_i	X		X	X	L↓
FN-3	N, D_i	X		X	X	P↓
VG-1	—	X	X	X	X	(NL) ↓
VG-2a	(NL)	X	X		X	Q↑
VG-2b	(NL)	X	X	X		Δ Ti↓
VG-3	(NL)	X		X	X	P↓

Table 2.1 Performance Evaluation Criteria [1]

Bergles et al [3] suggested a set of eight ($R1$-R_8) number of performance evaluation criteria as shown in Table 2.2.

Table 2.2 Performance Evaluation Criteria of Bergles et al [3]

		Criterion number							
		R_1	R_2	R_3	R_4	R_5	R_6	R_7	R_8
Fixed	Basic Geometry	×	×	×	×				
	Flow Rate	×						×	×
	Pressure Drop		×				×		×
	Pumping Power			×					
	Heat Duty				×	×	×	×	×
Objective	Increase Heat Transfer	×	×	×					
	Reduce pumping power				×				
	Reduce Exchange Size					×	×	×	×

It may be noted that FG-1a & FG-2a are similar to R_1 & R_3 respectively. Performance evaluation criteria R_1 have been used for present experimental work to determine heat transfer enhancement for different types of inserts.

Table 2.3 SUMMARIES OF IMPORTANT INVESTIGATIONS OF TWISTED TAPE IN LAMINAR FLOW [6]

SI No	Authors	Fluid	Configuration of twisted tape	Type of investigation	Observations	Comments
1	Saha and Dutta[8]	Water with (205< Pr < 518)	(a) Short length (b) Full length (c)Smoothly varying pitch (d)Regularly Spaced	Experiment in a circular tube	1) Friction and Nu low for short length tape (2) Short length tape requires small pumping power (3) Multiple twist and single twist has no difference on thermo hydraulic performance (4) Uniform pitch performs better then gradually decreasing pitch	It was observed that twisted tape is effective in laminar flow. Short length twisted tape perform better than full length tape.
2	Bergles and Hong [10]	Water (3<, Pr < 7) (83 < Re < 2460) Ethylene Glycol (84 < Pr< 192) (13 < Re < 390)	Full-length twisted tape	Experiment in circular tube	(1) Nu is function of twist ratio, Re and Pr (2) Friction is affected by tape twist only at high Re (3) Nu is 9 times that of empty tube	Twisted tape can been used as full-length twisted tape, half-length twisted tape and varying pitch twisted tape
4	Manglik and Bergles [12]	Water (3.5 < Pr < 6.5) and ethylene glycol (68 < Pr < 100)	Three different twist ratios: 3, 4.5 and 6	Experiment in isothermal tube	(1)Proposed correlation for friction and Nusselt number (2) Physical description of enhancement mechanisms	Pinching of twisted tape gives better results compared with connected thin rod

5	Saha et al. [13]	Fluids with 205 < Pr<518	Twisted tape (regularly spaced)	Experiment in circular tube	(1) Pinching of twisted tape gives better results than connecting thin rod for thermo hydraulic performance (2) Reducing tape width gives poor results; larger than zero phase angle not effective	
6	Lokanath and Misal [14]	Water (3 < Pr <6.5 and lube oil (Pr 418)	Twisted tape	Experiment in plate heat exchanger and shell and tube heat exchanger	(1) Large value of overall heat transfer coefficient produced in water-to water mode with oil-to water mode	
7	Lokanath [15]	Water (240 < Re < 2300) (2.6 < Pr < 5.4)	Full-length and half-length twisted tapes	Experimental in horizontal tube	(1) On unit pressure drop basis and on unit pumping power basis, half-length twisted tape is more effective than full-length twisted tape	
8	Liao and Xin[17]	(1) Water (2) Ethylene glycol (3) Turbine oil 5.5 < Pr < 590, 80 < Re< 50000	Segmented twisted tape and three-dimensional extended surfaces	Experiment in tube flow	(1) In a tube with three-dimensional Extended surfaces and twisted tape increases average Stanton number up to 5.8 times compared with empty smooth tube	

9	Ujhidy [18]	Water	Twisted tape	Experiment in channel	(1) Explained flow structure (2) Proved existence of secondary flow in tubes with helical static elements.	
10	Suresh Kumar [19]	Water	Twisted tape	Experiment in large diameter annulus	(1) Observed relatively large values of friction factor (2) Measured heat transfer in annulus with different configurations of twisted tapes	
11	Saha and Chakraborty [20]	Water (145 < Re<1480) (4.5 < Pr < 5.5)	Twisted tape (regularly spaced) (1.92<y<5.0)	Experiment in circular tube flow	(1) Larger number of turns may yield improved thermo hydraulic performance compared with single turn	
12	Saha and Bhunia [22]	Servotherm medium oil (205 < Pr < 512,45< Re < 840)	Twisted tape (twist ratio 2.5<y <10)	Experiment in circular tube	(1) Heat transfer characteristics depend on twist ratio, Re and Pr	Uniform pitch twisted tape performs better than gradually varying pitch twisted tape
13	Agarwal and Raja Rao[23]	Servotherm oil	Twisted tape	Experiment in circular tube	Nusselt number for augmented tube is more than plain tube	

CHAPTER 3

PRESENT EXPERIMENTAL WORK

3.1 SPECIFICATIONS OF HEAT EXCHANGER USED

The experiments were carried out on a double pipe heat exchanger with the specification listed below:-

Specifications of Heat Exchanger:

Inner pipe ID = 22mm

Inner pipe OD=25mm

Outer pipe ID =53mm

Outer pipe OD =61mm

Material of construction= Copper

Heat transfer length= 2.43m

Pressure tapping to pressure tapping length = 2.825m

Water at room temperature was allowed to flow through the inner pipe while hot water (set point 60°C) flowed through the annulus side in the counter current direction.

3.2 TYPES OF INSERTS USED

For experimental purpose eight type of inserts made from TMT rods of Dia. 8mm and 10mm were used.

1. TMT rod (without any baffle) of diameter 8mm.

2. TMT rod (without any baffle) of diameter 10mm.

3. TMT rod with baffles and baffle spacing 30cm (TMT rod dia. 8mm)

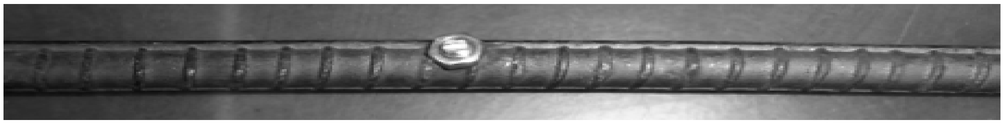
4. TMT rod with baffles and baffle spacing 20cm (TMT rod dia. 8mm)

5. TMT rod with baffles and baffle spacing 10cm (TMT rod dia. 8mm)

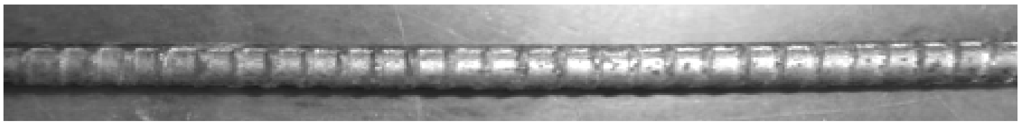
6. TMT rod with baffles and baffle spacing 30cm (TMT rod dia. 10mm)

7. TMT rod with baffles and baffle spacing 20cm (TMT rod dia. 10mm)

8. TMT rod with baffles and baffle spacing 10cm (TMT rod dia. 10mm)



10mm insert without any baffle



8mm insert without any baffle

Fig.3.1a

8mm insert with baffle, β=10cm

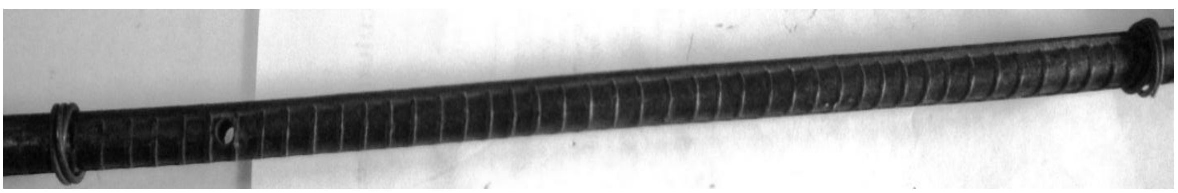
8mm insert with baffle, β=20cm

8mm insert with baffle, β=30cm

Fig. 3.1b

10mm insert with β=10cm

10mm insert with β=20cm



10mm insert with, β=30cm

Fig. 3.2

3.3 FABRICATION OF BAFFLES ON TMT RODS:

TMT rods of 8mm dia and 10 mm dia and length 2.94 meter were taken and four holes were drilled with equal spacing and with the help of nut and bolt the rods were supported inside the pipe. After leaving 5cm from both ends the rest 2.84 meter length was marked in 9 parts for 30 cm baffle spacing, 14 parts for 20 cm baffle spacing and similarly 28 parts for 10 cm baffle spacing. We used chalk for marking purpose and there after the marked space were twisted around with 1mm thickness GI wire that too in two rounds that worked as baffles.

3.4 EXPERIMENTAL SETUP:

Fig 3.3 shows the schematic diagram of the experimental setup. It's basically a double pipe heat exchanger consisting of an inner pipe of ID 22mm and OD 25mm, and an outer pipe of ID 53mm and OD 61 mm. the apparatus is also equipped with two rotameters for continuously measuring and maintaining the particular flow rate . There are two rotameter 1 for hot water flow measuring and another one for the cold water. There is an overhead cold water tank ie. source of cold water. There is another tank of capacity 500 litre which has an inbuilt heater and pump for providing hot of a particular temperature at a particular flow rate. We were also lucky to be equipped with the modern RTD meter. They have four different sensors situated at different locations to give four temperature T1, T2, T3, T4.

Hot water flow rate was kept constant at 1000 kg/hr. throughout the experiment. There is a U-Tube manometer for the pressure drop measurement it consist of two limbs well connected with the two points in the inner pipe. The fluid filled inside the manometer is Carbon Tetra Chloride (CCl_4).

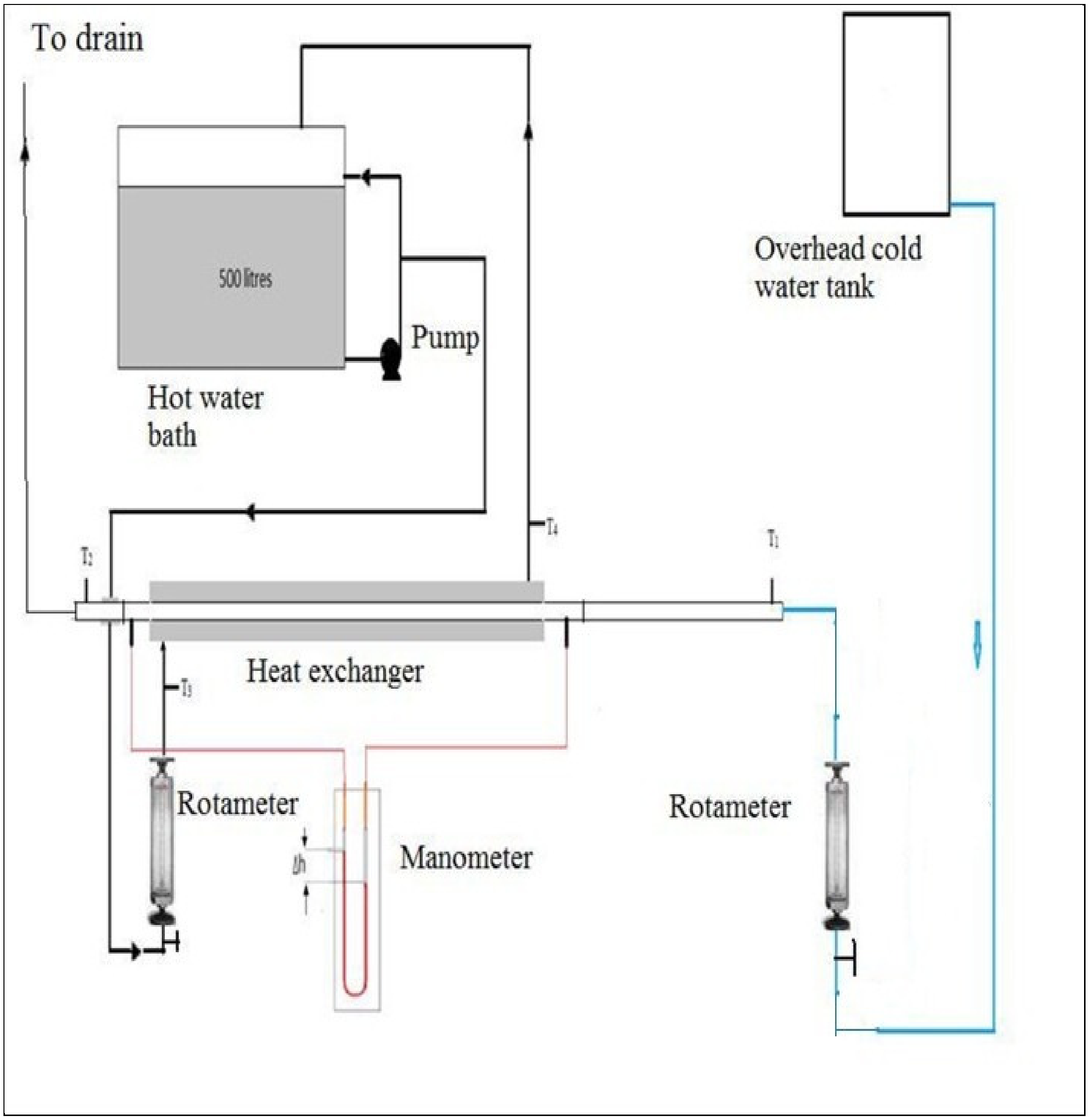


Fig 3.3 Schematic Diagram for the experimental setup

Fig 3.4 Photograph of the experimental setup

3.5 EXPERIMENTAL PROCEDURE:

1. All the RTD and Rotameter were calibrated first.

   i. For rotameter calibration we collected water in the bucket , weighted and simultaneously time was also noted. Thus mass flow rate was calculated.

   ii. We repeated this for three times for each particular reading and then took average of all. The readings are given in A.1.1 & A.1.2.

   iii. For RTD calibration, all the RTDs were simultaneously dipped in the same water bucket and readings were noted. T1 was made reference & corrections were made to other RTDs values (i.e. T2-T4) accordingly.

2. Standardization of the setup:

   Before starting the experimental study on friction & heat transfer in heat Exchanger using inserts, standardization of the experimental setup is done by obtaining the friction factor & heat transfer results for the smooth tube & comparing them with the standard equations available.

3. For friction factor determination:

   Pressure drop is measured for each flow rate with the help of manometer at room temperature.

   a. The U-tube manometer used carbon tetrachloride as the manometric liquid.

   b. Air bubbles were removed from the manometer so that the liquid levels in both limbs when the flow rate was zero.

   c. Water at room temperature is allowed to flow through the inner pipe of the heat exchanger.

   d. The manometer reading is noted.

4. For heat transfer coefficient calculation:
    a) Then, heater is put on to heat the water to 60°C in a constant temperature water tank of capacity 500 litres. The tank is provided with a centrifugal pump & a bypass valve for recirculation of hot water to the tank & to the experimental setup.
    b) Hot water at about 60°C is allowed to pass through the annulus side of heat exchanger at 1000KPH ($\mathbf{m}_h$=0.2778 Kg/sec).
    c) Cold water is now allowed to pass through the tube side of heat exchanger in counter current direction at a desired flow rate.
    d) The water inlet and outlet temperatures for both hot water & cold water (T_1-T_4) are recorded only after temperature of both the fluids attains a constant value.
    e) The procedure was repeated for different cold water flow rates ranging from 0.0601-0.3390 Kg/sec.
5. Preparation of Wilson chart:

$$\frac{1}{U_i} = \frac{1}{h_i} + \frac{h_i}{d_o \times h_o} + \frac{x_w \times d_i}{k_w \times d_l} + R_d \qquad (3.1)$$

where R_d is the dirt resistance

All the resistances, except the first term on the RHS of equation (1), are constant for this set of experiments.

For Re>10000, Seider Tate equation for smooth tube is of the form: $h_i = A \times Re^{0.8}$

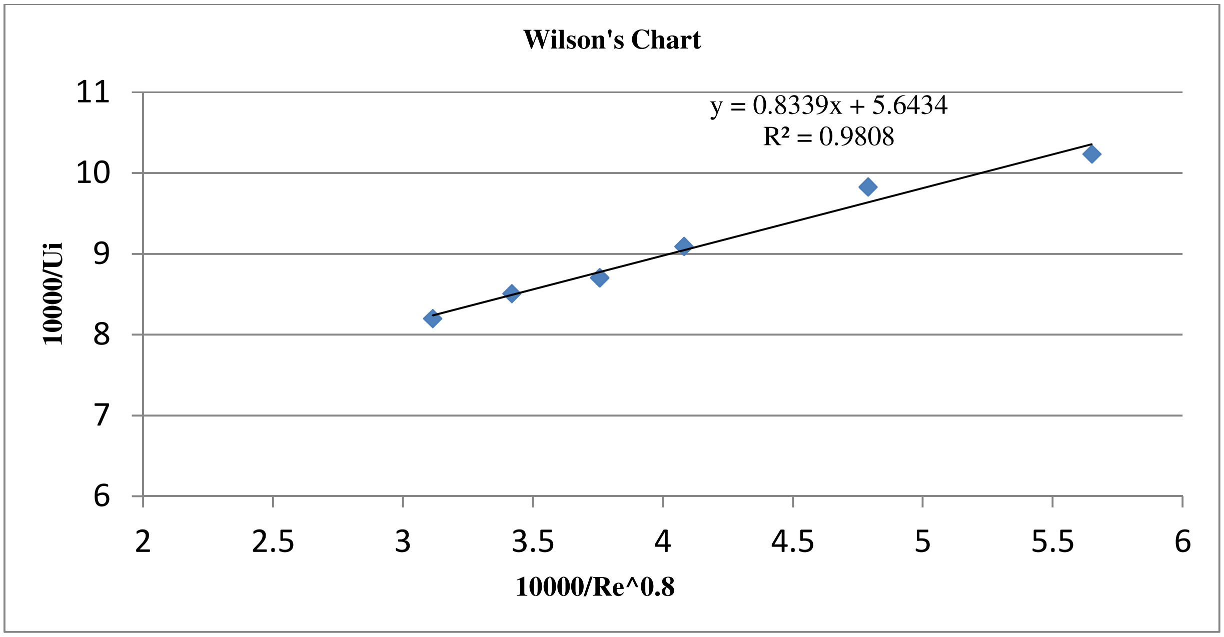


Fig 3.5

Therefore Eq. (3.1) can be written as

$$\frac{1}{U_i} = \frac{1}{A * Re^{0.8}} + K$$

K is to be found from the Wilson chart ($1/U_i$ vs. $1/Re^{0.8}$) as the intercept on the y-axis.

$K = 5.6434*10^{-4}$

6. After confirmation of validity of experimental values of friction factor & heat transfer coefficient in smooth tube with standard equations, friction factor & heat transfer studies with inserts were conducted.

7. The friction factor & heat transfer observations & results for all the cases are presented in Tables A.2.1-A.2.9 & A.3.1-A.3.9 respectively.

3.6 STANDARD EQUATIONS USED:

I. Friction factor (f0) calculations:

a. For Re< 2100

$$f = \frac{16}{Re} \tag{3.3}$$

b. For Re>2100

Colburn's Equation:

$$f = \frac{0.046}{Re^{0.2}} \tag{3.4}$$

II. Heat transfer calculations

i. Laminar Flow:

For Re<2100

Nu= f(Gz)

Where $Gz = \frac{Re \times Pr \times d_i}{L}$ (3.5)

a. For Gz<100, Hausen Equation is used.

$$Nu = 3.66 + \frac{0.085Gz}{1+0.045Gz^{0.67}} \left(\frac{\mu b}{\mu w}\right)^{0.14} \tag{3.6}$$

b. For Gz>100, Seider Tate equation is used.

$$Nu = 1.86Gz^{1/3}\left(\frac{\mu_b}{\mu_w}\right)^{0.14} \tag{3.7}$$

ii. Transition Zone:

For 2100<Re<10000, Hausen equation is used

$$Nu = 0.116\left(Re^{2/3} - 125\right) \times Pr^{1/3} \times \left(1 + \left(\frac{D}{l}\right)^{2/3}\left(\frac{\mu_b}{\mu_w}\right)^{0.14}\right. \tag{3.8}$$

iii. Turbulent Zone:

For Re>10000, Seider-Tate equation is used.

$$Nu = 0.023 \times Re^{0.8} \times Pr^{1/3} \times \left(\frac{\mu_b}{\mu_w}\right)^{0.14} \tag{3.9}$$

Viscosity correction Factor $\left(\frac{\mu_b}{\mu_w}\right)^{0.14}$ is assumed to be equal to 1 for all calculations as this value for water in present case will be very close to 1 & the data for wall temperatures is not measured.

CHAPTER 4

SAMPLE CALCULATIONS

4.1 ROTAMETER CALIBRATION:

For 600 Kph (Table No. A1.1)

Observation No.1

Weight of water collected=1.97 kg

Time=13.51sec

$\mathbf{m_1}$=0.1458 kg/sec

Observation No.2

Weight of water collected=1.8 kg

Time=11.26 sec

$\mathbf{m_2}$=0.1598 kg/sec

Observation No.3

Weight of water collected=1.76 kg

Time=11.05 sec

$\mathbf{m_3}$=0.1593 kg/sec

$$\mathrm{m} = \frac{\mathrm{m_1} + \mathrm{m_2} + \mathrm{m_3}}{3} = \frac{0.1458 + 0.1598 + 0.1593}{3} = 0.1550 \text{ Kg/sec}$$

Diff.= 7%

4.2 PRESSURE DROP & FRICTION FACTOR CALULATIONS:

For 8 mm insert with β=30cm (Table No.A2.4)

m=0.1550 Kg/sec

Experimental friction factor:

$$Area = \frac{\pi * {d_i}^2}{4} = \frac{\pi * 0.022^2}{4} = 3.8 * 10^{-4} m^2$$

$$v = \frac{m}{A * \rho w} = \frac{0.1550}{3.8 * 10^{-4} * 1000} = 0.41 m/sec$$

$$\Delta P = (\rho_{ccl4} - \rho_{H2o}) * g * \Delta h = (1603 - 1000) * 9.81 * 0.38 = 2254\ N/m^2$$

$$f_a = \frac{\Delta P * d_i}{2 * \rho * L * v^2} = \frac{2254 * 0.022}{2 * 1000 * 2.83 * 0.41^2} = 0.053$$

For viscosity calculation:

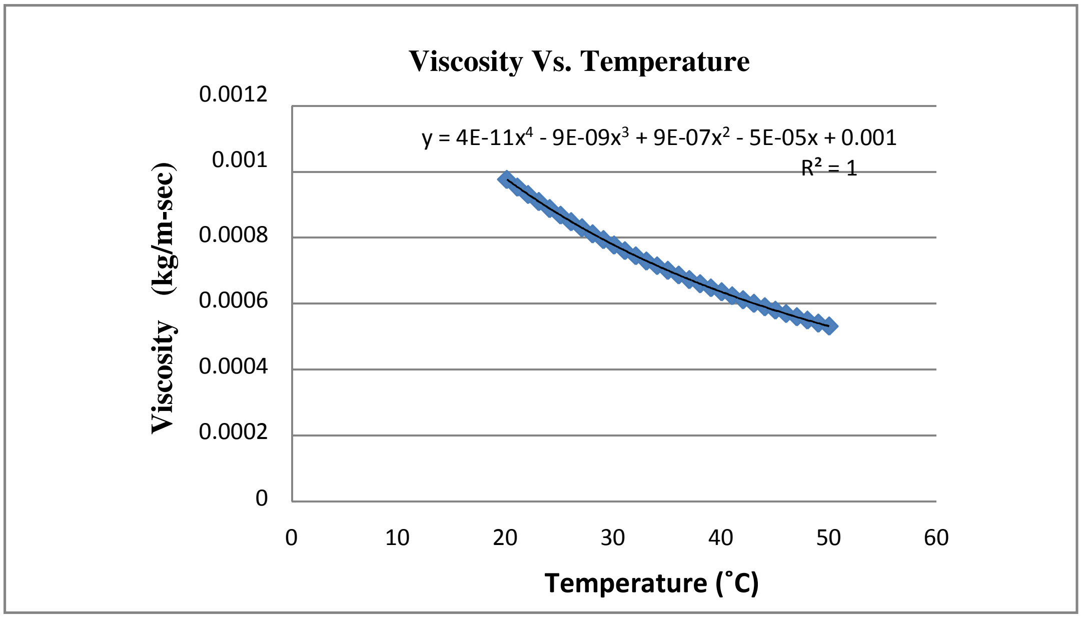


Fig 4.1 Viscosity vs. Temperature

$$\mu = 4\times10^{-11}T^{4}-9\times10^{-09}T^{3}+9\times10^{-07}T^{2}-5\times10^{-05}T+0.0017 \quad (4.1)$$

Theoretical friction factor calculation for smooth tube:

$$\mathrm{Re} = \frac{4 * \mathrm{m}}{\pi * \mathrm{d_i} * \mu} = \frac{4 * 0.1550}{\pi * 0.022 * 0.00084} = 10639$$

$f_o = 0.046 \times Re^{-0.2} = 0.046 \times 10639^{-0.2} = 7.2\times10^{-3}$

$$\frac{f_a}{f_o} = \frac{0.053}{0.0072} = 7.34$$

4.3 HEAT TRANSFER COEFFICIENT CALCULATION:

For 8mm insert with β=30cm (Table No.A3.4)

$\mathbf{m_c}$ = 0.1550 kg/sec (600Kph) & $\mathbf{m_h}$=0.2778 kg/sec

NOTE: Temperature correction has already been taken into account while giving data in appendix.

$T_1 = 26.8\ ^0C$

$T_2 = 36.1\ ^0C$

$T_3 = 68.4\ ^0C$

$T_4 = 62.9\ ^0C$

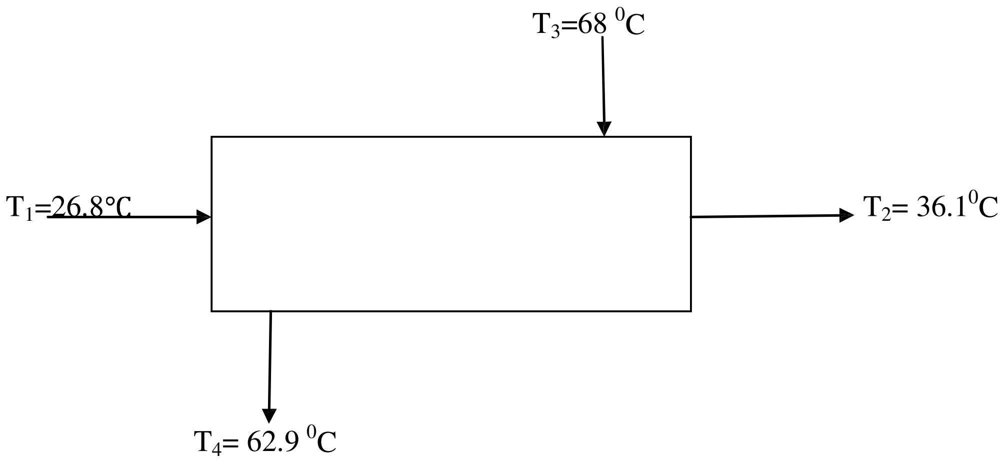


Fig. 4.2 Temperature in different RTDs

$\Delta T_1 = T_4 - T_1 = (62.9-26.8) \quad = \quad 36.1\ ℃$

$\Delta T_2 = T_3 - T_2 = (63.4-36.1) \quad = \quad 32.3\ ℃$

$$\mathrm{LMTD} = \frac{\Delta T_1 - \Delta T_2}{\ln \frac{\Delta T_1}{\Delta T_2}} = \frac{36.1 - 32.3}{\ln \frac{36.1}{32.3}} = 34.16℃$$

$Q_1 = m_c \times C_{pc} \times (T_2 - T_1) = 0.1550 \times 4187 \times (36.1-26.8) = 6036\ W$

$Q_2 = m_h \times C_{ph} \times (T_3 - T_4) = 0.2778 \times 4187 \times (68.4-62.9) = 6397\ W$

$$\text{Heat balance error} = \frac{6036 - 6397}{6397} * 100 = -5.6\%$$

$$Q_{avg} = \frac{Q_1 + Q_2}{2} = 6216\ W$$

$$\text{Heat transfer area} = \pi * d_i * l = \pi * 0.022 * 2.43 = 0.1680 m^2$$

$$U_i = \frac{Q_{avg}}{A_i * LMTD} = \frac{6216}{0.1680 * 34.16} = 1083 W/m^2℃$$

$$Re = \frac{4 * m}{\pi * d_i * \mu} = \frac{4 * 0.1550}{\pi * 0.022 * 0.00078} = 11546$$

h_i can be calculated using Eq. (3.1)

$$\frac{1}{U_i} = \frac{1}{h_i} - K \qquad (4.2)$$

K is found from the Wilson chart ($1/U_i$ vs. $1/Re^{0.8}$) as the intercept on the y-axis.

$K = 5.6434 \times 10^{-4}$ (Refer Fig 3.7)

$$\frac{1}{h_i} = \frac{1}{U_i} - K = \frac{1}{1083} - 5.6434 * 10^{-4}$$

$\Rightarrow h_a = 2788 \text{ W/m}^2\,{}^\circ\text{C}$

Theoretical Calculation for smooth tube

$$\text{Nu} = 0.023 \times \text{Re}^{0.8} \times \text{Pr}^{1/3}$$

$$\Rightarrow \frac{h_i \times d_i}{k} = 0.023 \times \text{Re}^{0.8} \times \text{Pr}^{1/3}$$

$$\Rightarrow h_i = \frac{0.023 \times k}{d_i} \times \text{Re}^{0.8} \times \text{Pr}^{1/3} \qquad (4.3)$$

For Prandtl Number calculation:

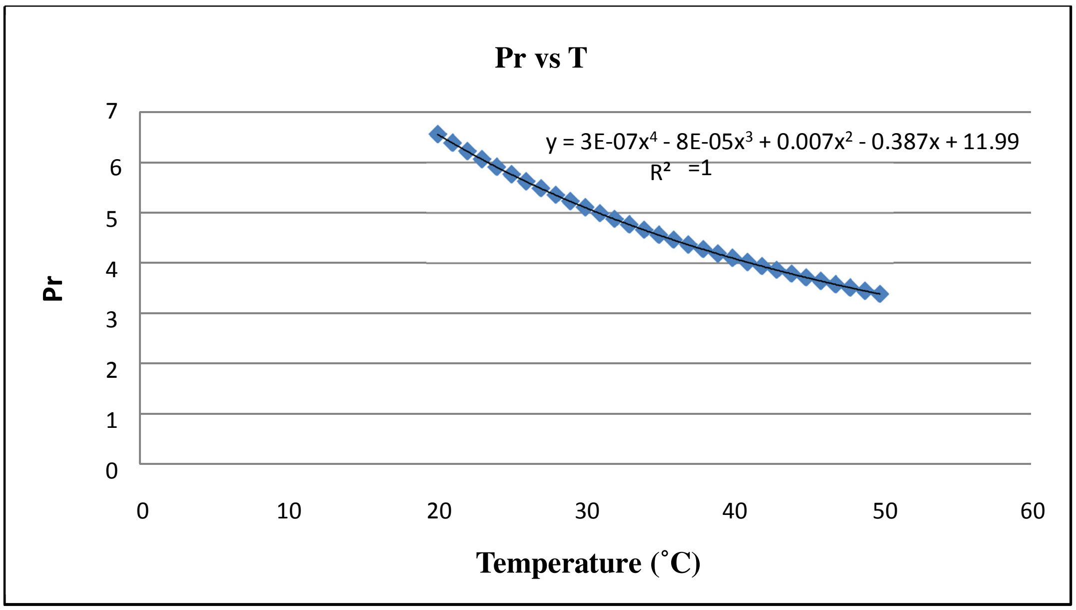


Fig 4.3 Prandtl Number vs. Temperature

$$\text{Pr}=3\times10^{-07}T^4-8\times10^{-05}T^3+0.0072\times T^2-0.3873\times T+11.995 \qquad (4.4)$$

$$T_{avg} = \frac{T_1 + T_2}{2} = 31.45°\text{C}$$

Pr (at $T=T_{avg}$) =4.74

$$h_o(h_i \text{ for smooth tube}) = \frac{0.023 * 0.6322}{0.022} * 11546^{0.8} * 4.74^{1/3}$$

$h_0 = 1975 \text{ W/m}^2\,{}^\circ\text{C}$

$$R_1 = \frac{h_a}{h_o} = 1.41$$

CHAPTER 5

RESULTS & DISCUSSION

5.1 <u>FRICTION FACTOR RESULTS:</u>

All friction factor results and f_a/f_o values of all the cases are tabled in the tables A.2.1-A.2.9. In almost all Reynolds no. range(neglecting low values of Reynolds no.) the difference of f_{exp} and f_{theo} is very much within the $\pm 10\%$. That validated the equation we used for our experimental purpose.

As the ΔH values were very small (0.1-0.8cm) for low Re & the manometer's least count was 0.1cm, so we cannot measure those low pressure drops with higher accuracy.

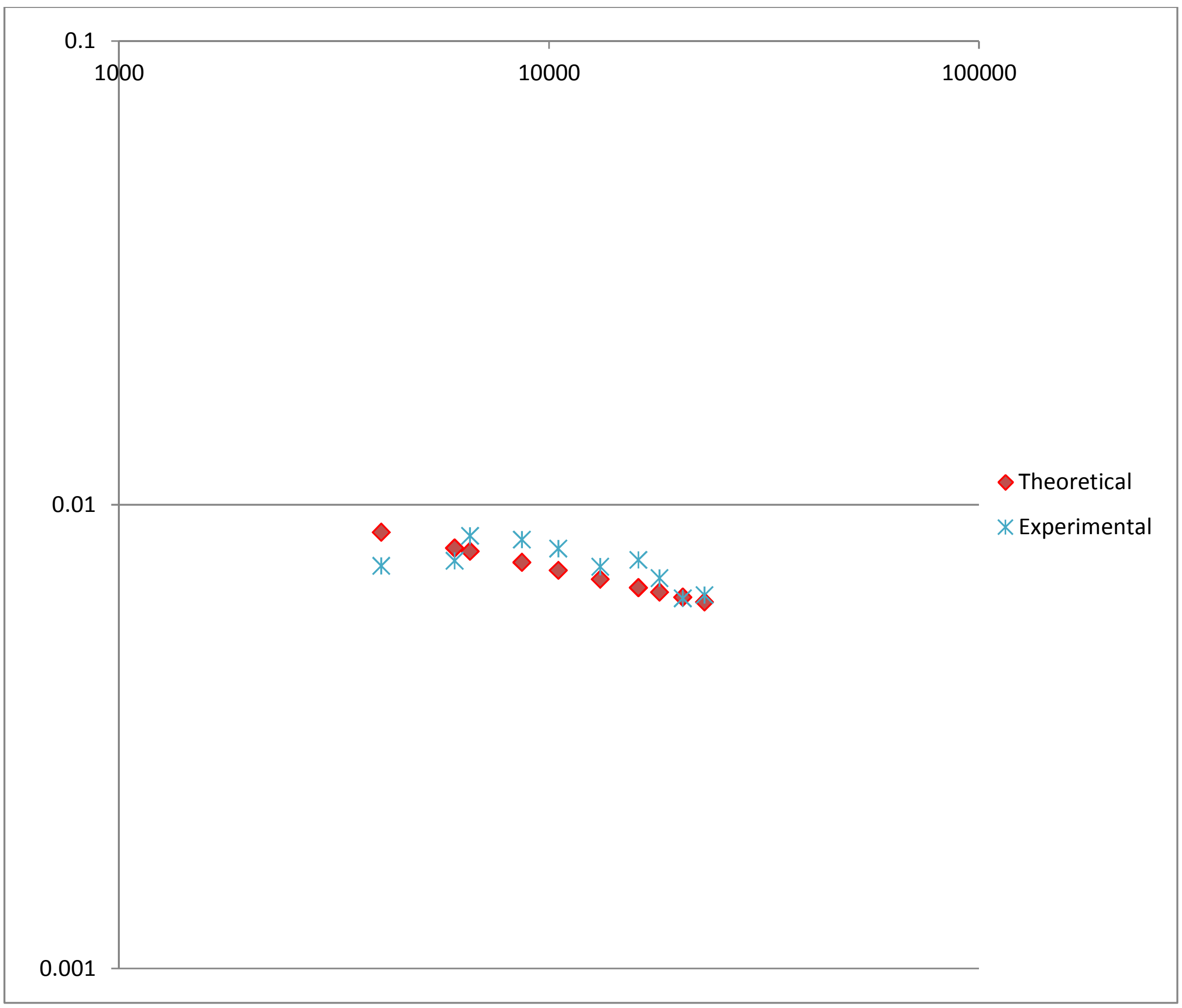


Fig 5.1 Friction Factor vs. Reynolds number for Smooth Tube

Fig. 5.2 represents the variation of friction factor with Reynolds no. for 8mm insert without baffle, with baffles of β=10, 20, 30 cm and for 10mm insert without baffle, with baffles of β=10, 20, 30 cm. as the number of baffles increases the friction factor also follows the same pattern. So for 10mm insert with β=10cm friction factor is highest . Inserts with baffles are giving high friction factor because of increase in the degree of turbulence.

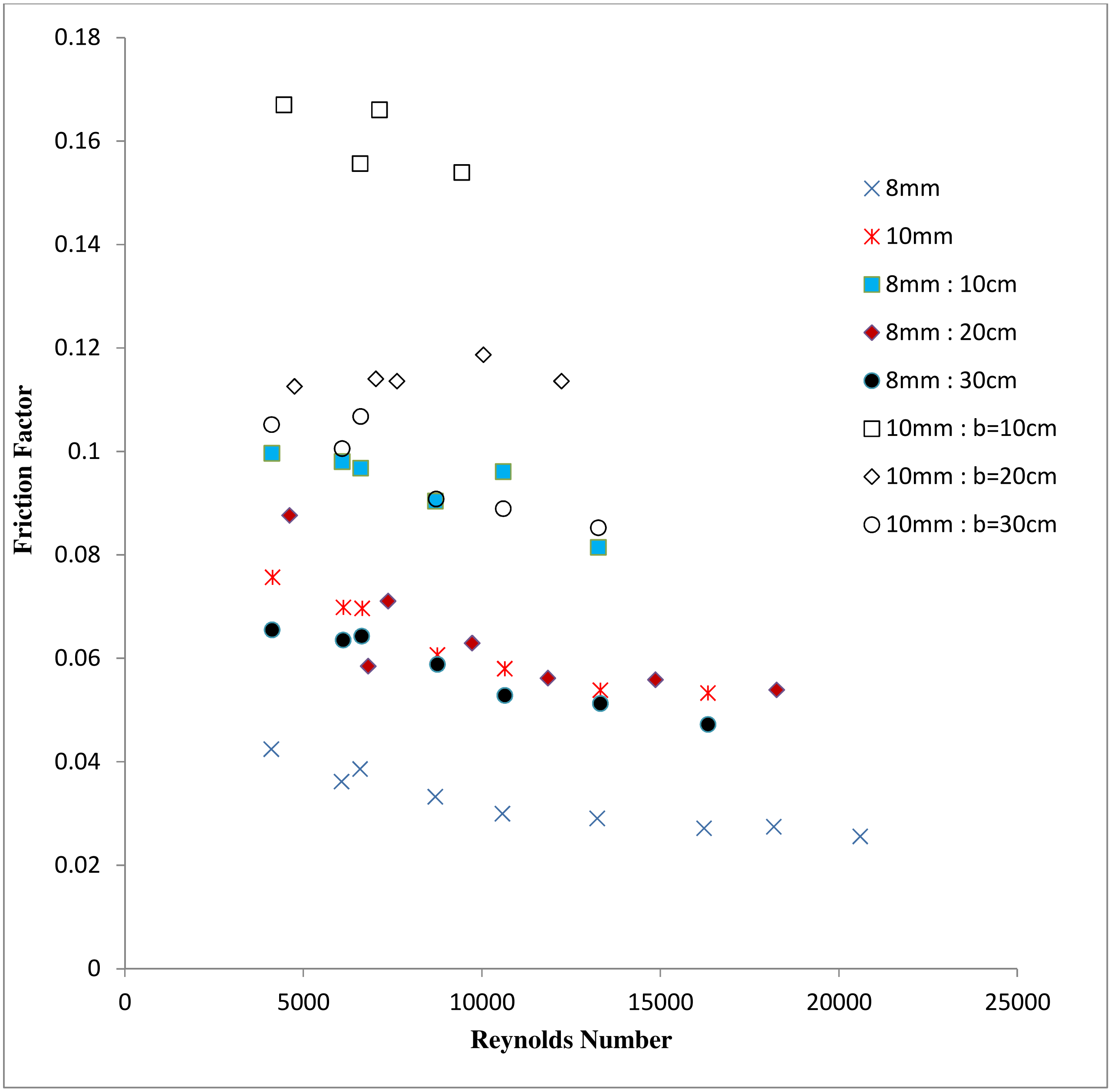


Fig 5.2 Friction factor vs. Reynolds number for Smooth tube, inserts with baffles or without baffles.

Fig 5.3 shows the variation of f_a/f_o with Reynolds number for 8mm inserts with or without baffles and also for the 10mm inserts with or without baffles.

a. f_a/f_o is found to be highest for 10mm insert with β=10cm.

b. f_a/f_o is lowest in case of 8mm insert without any baffle.

c. f_a/f_o is large for all 10mm inserts with baffles.

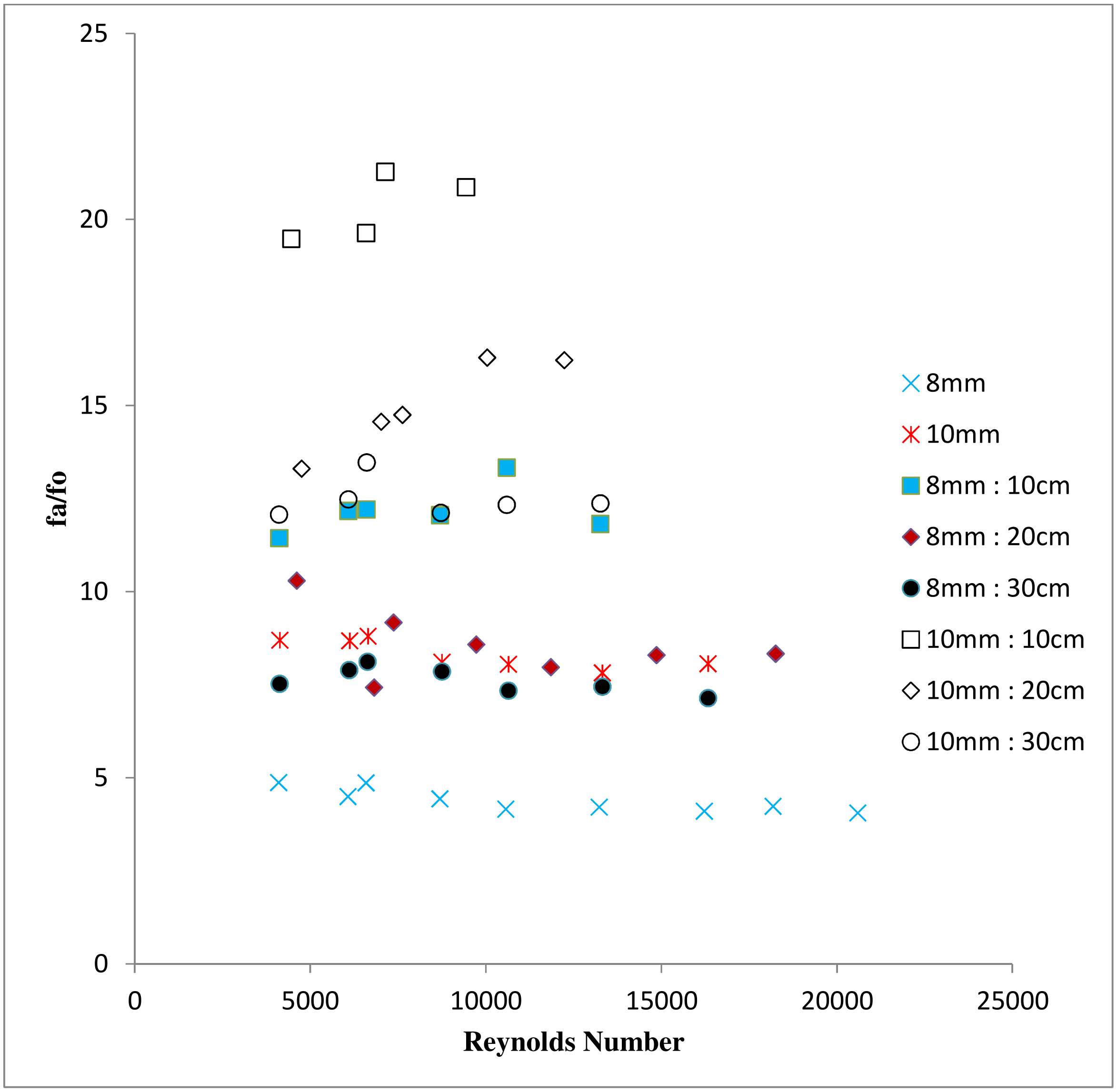


Fig 5.3 f_a/f_o vs. Reynolds Number for 8mm inserts with or without baffles and 10mm inserts with or without baffles.

5.2 <u>HEAT TRANSFER COEFFICIENT RESULTS:</u>

Table A.3.1-A.3.9 gives the heat transfer results for smooth tube, 8mm insert without any baffle and with baffles(β=10, 20, 30) and for 10mm insert without any baffle and also with baffles(β=10, 20, 30) along with the corresponding performance evaluation criteria R_1 for each of the readings. As shown in fig.5.4, the difference between h_{exp} & h_{theo} is very low. So that unanimously validates our heat equations for the experimental setup. We have neglected the higher deviation between h_{exp} & h_{theo} for low Reynolds number because this can be attributed to the phenomenon of natural convection taking place along with forced convection that is negligible in comparison to forced convection at for high Reynolds no.

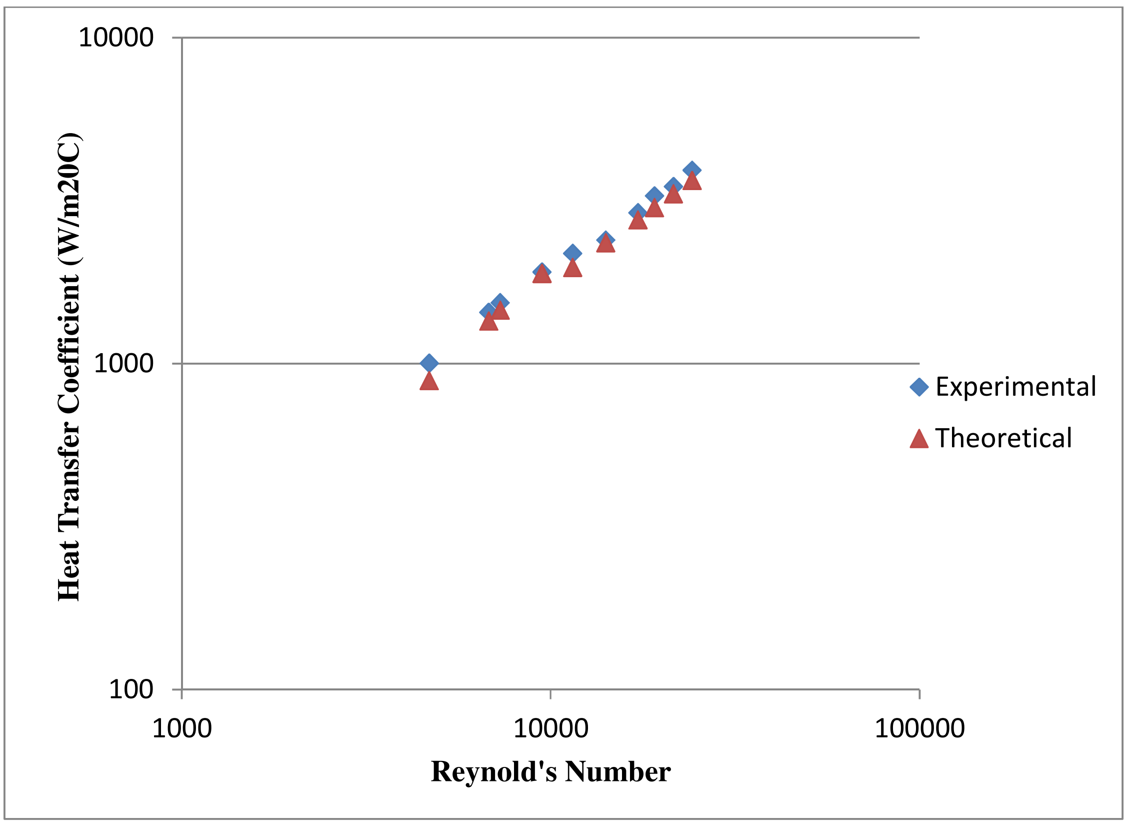


Fig 5.4 Heat transfer coefficient vs. Reynolds Number for smooth tube

Fig 5.5 represents the variation in heat transfer coefficient (h_a) with Reynolds no. for Smooth tube, 8mm insert without any baffle and with baffles(β=10, 20, 30) and for 10mm insert without any baffle and also with baffles(β=10, 20, 30) . As the baffle spacing (β) decreases a higher degree of turbulence is created & hence the heat transfer coefficient increases as the baffle spacing decreases.

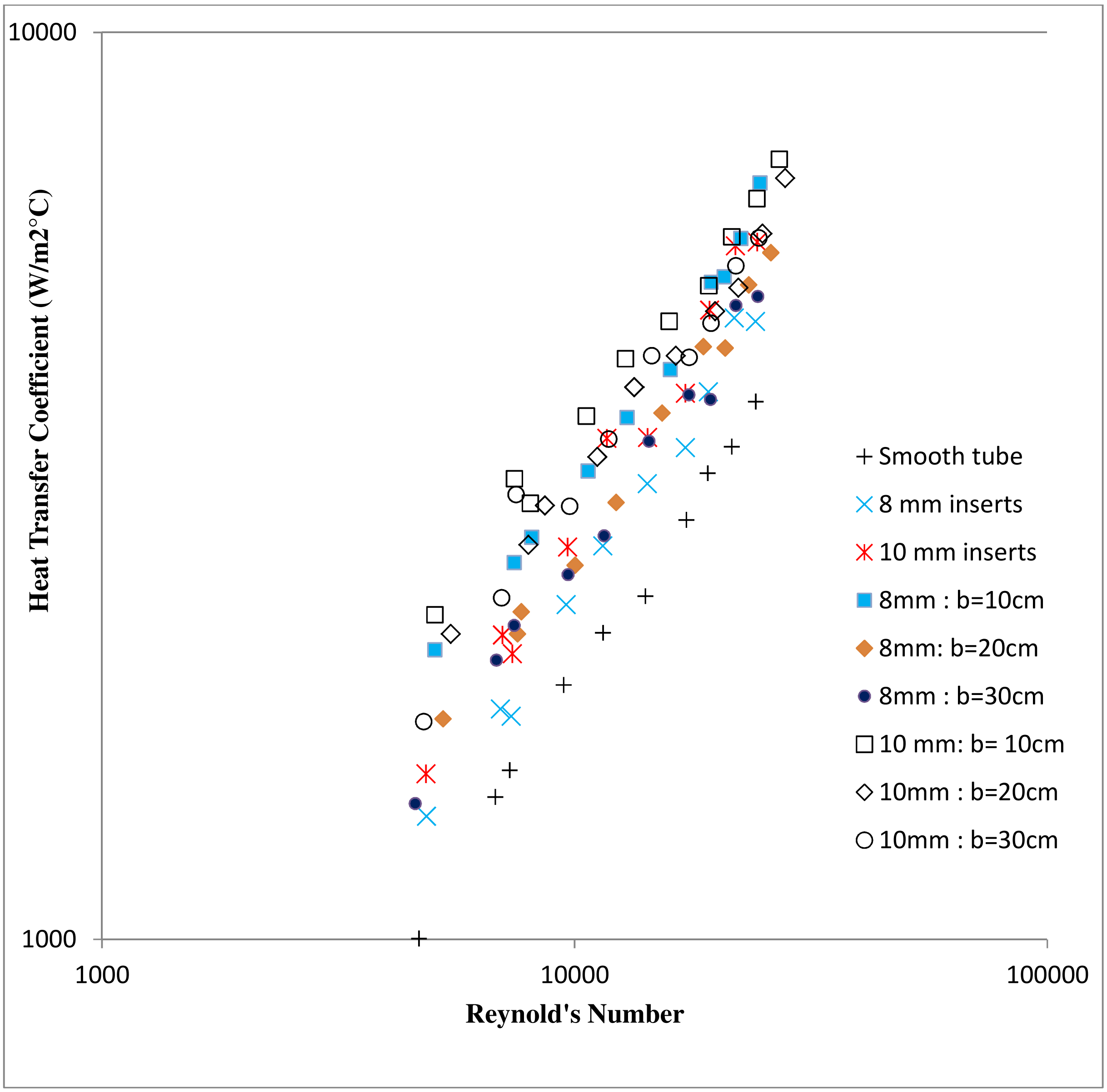


Fig 5.5 Heat transfer coefficient vs. Reynolds Number for Smooth tube, inserts with or without baffles.

In fig. 5.6, a plot between performance evaluation criteria R1 Vs. Reynolds no. is shown.

Maximum value of R1 is observed for 10mm insert (β=10cm). From this we can conclude that this is the best arrangement out of all arrangements tested for this experiment.

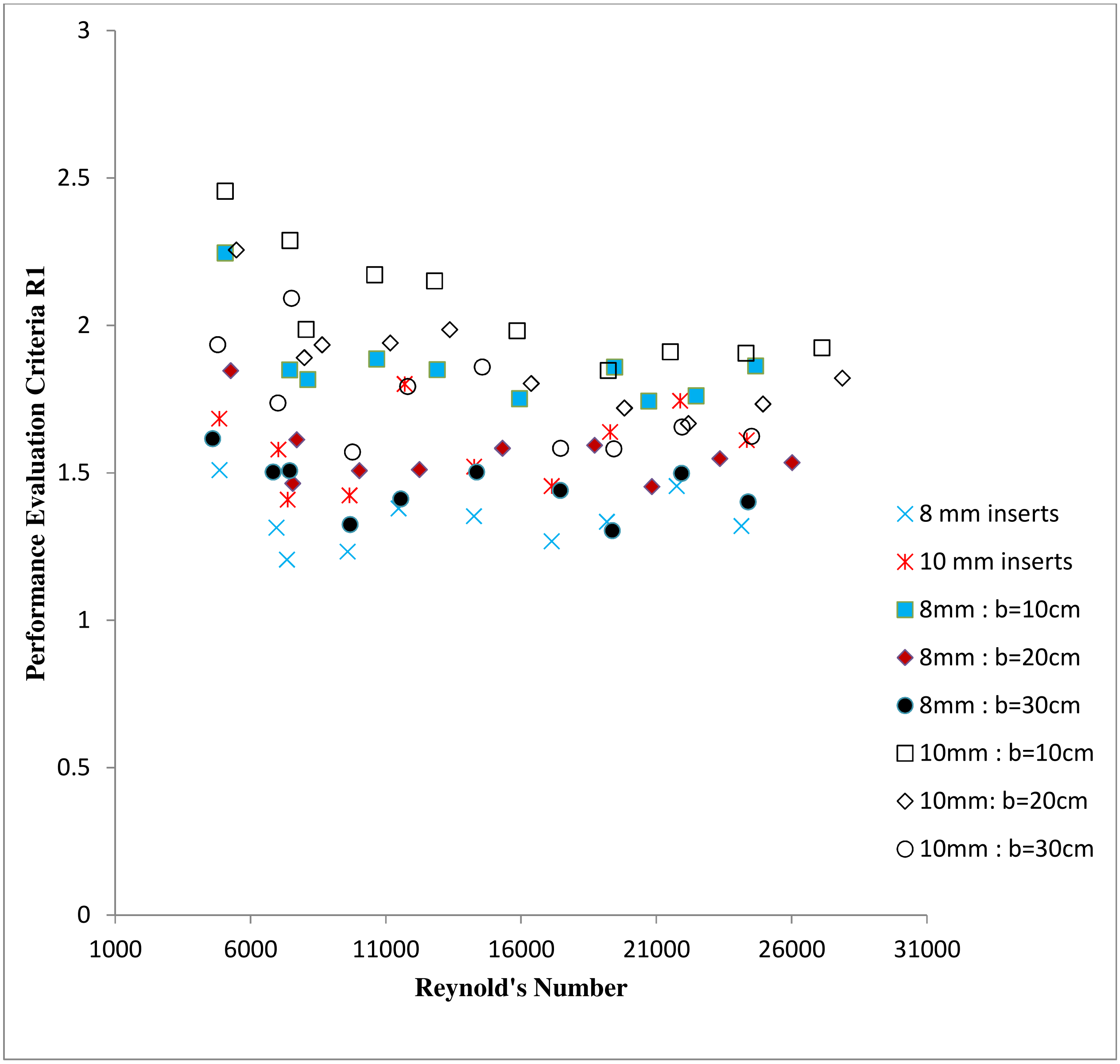


Fig 5.6 Performance evaluation criteria, R_1 vs. Reynolds Number for inserts with or without baffles.

CHAPTER 6

CONCLUSION

The range of Performance evaluation criteria R_1 (based on constant mass flow rate), & fa/fo for different inserts used is given below:

Table 6.1 Range of R_1, f_a/f_o for different inserts:

Sl. no.	Insert	Range of R1	Range of fa/fo
1	8mm	1.21-1.51	4.05-4.87
2	10mm	1.41-1.80	7.82-8.80
3	8mm : β= 30cm	1.30-1.62	7.14-8.12
4	8mm : β= 20cm	1.45-1.85	7.43-10.30
5	8mm : β = 10cm	1.74-2.25	11.44-13.34
6	10mm : β = 30cm	1.57-2.09	12.07-13.48
7	10mm : β = 20cm	1.67-2.26	13.30-16.30
8	10mm : β = 10cm	1.85-2.46	19.48-21.29

1. For same baffle spacing (β), 8mm & 10mm inserts with baffles shows greater heat transfer coefficient & friction factor than the value we get for inserts without baffles, because of increased degree of turbulence created.
2. On the basis of performance evaluation criteria R_1, we can say that the 10mm insert with baffle spacing (β=10cm) gives the highest R1 range with the maximum value of Heat transfer coefficient around 2.46 times of the value for the smooth tube.
3. From the table 6.1, we can easily infer that the effect of 10mm insert (without baffles) and 8mm insert with β = 30cm are almost equivalent on both the performance evaluation criteria R1 & fa/fo.

4. With decrease in baffle spacing (β), heat transfer coefficient increases but at the same time pressure drop also increases.

CHAPTER 7

SCOPE FOR FUTURE WORK

Further modification can be done using this study as base. Some of the possibilities are mentioned below:

1. Distance between two consecutive baffle (baffle spacing) can be varied and their effect on heat transfer coefficient and friction factor can easily be noted down.

2. Pressure drop is a big loss of this modification so studies can be made to minimize the pressure drop.

3. Design of baffle are also a subject to affect both the friction factor and heat transfer coefficient.

4. The same experiment can also be tested with cooling operations.

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APPENDIX

A.1. CALIBRATION

A.1.1 ROTAMETER CALIBRATION

Rotameter readings (kg/hr)	Observation 1			Observation 2			Observation 3					
	Wt. (Kg)	Time (Sec)	M (Kg/s)	Wt. (Kg)	Time (Sec)	M (Kg/s)	Wt. (Kg)	Time (sec)	M (Kg/s)	Average (Kg/sec)	Actual flow (Kg/hr)	%age diff.
300	0.71	11.59	0.0613	0.71	11.76	0.060	0.72	12.18	0.059	0.0601	216.36	27.88
350	1.13	12.68	0.0891	1.04	11.75	0.0885	1.13	12.70	0.0890	0.0889	320.40	8.46
400	1.65	17.48	0.0944	1.19	12.19	0.0976	1.44	14.70	0.0979	0.0966	347.76	13.06
500	1.21	9.70	0.1247	1.06	8.24	0.1286	1.63	12.61	0.1293	0.1275	459	8.20
600	1.97	13.51	0.1458	1.80	11.26	0.1598	1.76	11.05	0.1593	0.1550	558	7.0
750	2.18	11.24	0.1939	2.0	10.44	0.1916	2.03	10.33	0.1965	0.1940	698.40	6.88
900	2.63	11.06	0.2378	2.55	10.74	0.2374	2.57	10.77	0.2386	0.2379	856.44	4.84
1000	2.80	10.73	0.2610	2.90	10.74	0.2700	2.91	10.84	0.2685	0.2665	959.40	4.06
1100	2.85	9.70	0.2938	3.10	10.18	0.3045	3.28	10.66	0.3077	0.3020	1087.20	1.16
1250	3.39	10.02	0.3383	3.34	9.93	0.3363	3.68	10.75	0.3423	0.3390	1220.40	2.37

A.1.2 RTD CALIBRATION:

Sl No	Temperature Readings			
	T1	T2	T3	T4
1	20.7	20.9	20.5	20.7
2	20.8	21.0	20.6	20.8
3	20.7	20.9	20.5	20.7
4	20.7	20.9	20.5	20.7
Correction	0	-0.2	+0.2	0

A.2. <u>FRICTION FACTOR RESULTS:</u>

A.2.1 STANDARDISATION OF SMOOTH TUBE (f vs. Re)

m (Kg/sec)	ΔH (cm)	T (˚C)	Re	ΔP(N/m^2)	f_{exp}*1000	f_{theo}*1000	%diff
0.0601	0.80	26.80	4073	47.32	7.38	8.73	15.42
0.089	1.80	26.80	6032	106.5	7.57	8.07	6.13
0.0966	2.40	26.80	6547	141.9	8.56	7.94	-7.99
0.1275	4.10	26.80	8641	242.5	8.40	7.51	-11.95
0.155	5.80	26.80	10505	343.1	8.04	7.22	-11.43
0.194	8.30	26.80	13148	491	7.35	6.90	-6.46
0.2379	12.9	26.80	16124	763.1	7.59	6.63	-14.61
0.2665	14.8	26.80	18062	875.5	6.94	6.48	-7.19
0.302	17.2	26.80	20468	1017	6.28	6.32	0.54
0.339	22.0	26.80	22975	1301	6.38	6.17	-3.33

A.2.2 FRICTION FACTOR vs. Re FOR 8mm INSERTS

m (Kg/sec)	ΔH(cm)	T(˚C)	Re	ΔP(N/m^2)	fa*1000	fo*1000	fa/fo
0.0601	4.6	27.10	4099	272.0	42.43	8.71	4.87
0.089	8.6	27.10	6070	508.7	36.18	8.06	4.49
0.0966	10.8	27.10	6589	638.9	38.56	7.92	4.87
0.1275	16.2	27.10	8696	958.3	33.20	7.50	4.43
0.155	21.6	27.10	10572	1278	29.96	7.21	4.15
0.194	32.8	27.10	13232	1940	29.04	6.89	4.21
0.2379	46.1	27.10	16226	2727	27.14	6.62	4.10
0.2665	58.4	27.10	18177	3455	27.40	6.47	4.24
0.302	70.0	27.10	20598	4141	25.57	6.31	4.05

A.2.3 FRICTION FACTOR vs. Re FOR 10mm INSERTS

m (Kg/sec)	ΔH(cm)	T(˚C)	Re	ΔP(N/m^2)	f_a*1000	f_o*1000	f_a/f_o
0.0601	8.2	27.50	4134	485.07	75.64	8.70	8.69
0.089	16.6	27.50	6121	982.0	69.83	8.04	8.68
0.0966	19.5	27.50	6644	1154	69.63	7.91	8.80
0.1275	29.6	27.40	8751	1751	60.67	7.49	8.10
0.155	41.8	27.40	10639	2473	57.97	7.20	8.05
0.194	60.8	27.40	13316	3597	53.83	6.88	7.82
0.2379	90.5	27.40	16329	5353	53.28	6.61	8.06

A.2.4 FRICTION FACTOR vs. Re FOR 8mm INSERTS WITH β = 30cm

m (Kg/sec)	ΔH(cm)	T(˚C)	Re	ΔP(N/m^2)	f_a*1000	f_o*1000	f_a/f_o
0.0601	7.1	27.40	4125	420.0	65.49	8.70	7.52
0.089	15.1	27.40	6109	893.23	63.52	8.05	7.89
0.0966	18.0	27.40	6630	1065	64.27	7.92	8.12
0.1275	28.7	27.40	8751	1698	58.82	7.49	7.86
0.155	38.1	27.40	10639	2254	52.84	7.20	7.34
0.194	57.9	27.40	13316	3425	51.26	6.88	7.45
0.2379	80.2	27.40	16329	4744	47.22	6.61	7.14

A.2.5 FRICTION FACTOR vs. Re FOR 8mm INSERTS WITH β = 20cm

m (Kg/sec)	ΔH(cm)	T(˚C)	Re	ΔP(N/m^2)	f_a*1000	f_o*1000	f_a/f_o
0.0601	9.5	33.0	4612	562.0	87.63	8.51	10.29
0.089	13.9	32.9	6817	822.2	58.47	7.87	7.43
0.0966	19.9	32.7	7371	1177	71.06	7.75	9.17
0.1275	30.7	32.7	9728	1816	62.92	7.33	8.58
0.155	40.5	32.8	11849	2396	56.17	7.05	7.97
0.194	63.1	32.9	14859	3733	55.86	6.74	8.29
0.2379	91.5	33.0	18256	5413	53.87	6.46	8.33

A.2.6 FRICTION FACTOR vs. Re FOR 8mm INSERTS WITH β = 10cm

m (Kg/sec)	ΔH(cm)	T(˚C)	Re	ΔP(N/m^2)	f_a*1000	f_o*1000	f_a/f_o
0.0601	10.8	27.3	4116	638.9	73.80	8.71	11.44
0.089	23.3	27.2	6083	1378	78.66	8.05	12.17
0.0966	27.1	27.2	6603	1603	97.48	7.92	12.21
0.1275	44.1	27.1	8696	2609	90.39	7.50	12.06
0.155	69.3	27.2	10594	4099	100.8	7.21	13.34
0.194	92.0	27.2	13260	5442	81.45	6.89	11.82

A.2.7 FRICTION FACTOR vs. Re FOR 10mm INSERTS WITH β = 30cm

m (Kg/sec)	ΔH(cm)	T(˚C)	Re	ΔP(N/m^2)	f_a*1000	f_o*1000	f_a/f_o
0.0601	11.4	27.2	4108	674.4	105.2	8.71	12.07
0.089	23.9	27.2	6083	1414	101.0	8.05	12.48
0.0966	29.9	27.2	6603	1769	106.8	7.92	13.50
0.1275	44.3	27.2	8715	2621	90.80	7.49	12.12
0.155	64.1	27.2	10594	3792	88.90	7.21	12.34
0.194	96.3	27.2	13260	5697	85.26	6.89	12.37

A.2.8 FRICTION FACTOR vs. Re FOR 10mm INSERTS WITH β = 20cm

m (Kg/sec)	ΔH(cm)	T(˚C)	Re	ΔP(N/m^2)	f_a*1000	f_o*1000	f_a/f_o
0.0601	12.2	34.6	4751	721.7	112.5	8.46	13.30
0.089	27.1	34.5	7023	1603	114.0	7.82	14.57
0.0966	31.8	34.5	7623	1881	113.5	7.70	14.75
0.1275	57.9	34.4	10043	3425	118.7	7.28	16.29
0.155	81.9	34.5	12231	4845	113.6	7.00	16.22

A.2.9 FRICTION FACTOR vs. Re FOR 10mm INSERTS WITH β = 10cm

m (Kg/sec)	ΔH(cm)	T(˚C)	Re	ΔP(N/m^2)	f_a*1000	f_o*1000	f_a/f_o
0.0601	18.1	31.2	4455	1071	166.9	8.57	19.48
0.089	37.0	31.1	6584	2189	155.6	7.93	19.64
0.0966	46.5	31.0	7133	2751	166.0	7.80	21.28
0.1275	75.1	31.1	9433	4442	153.9	7.38	20.87

A.3. HEAT TRANSFER RESULTS:

A.3.1 STANDARDISATION OF SMOOTH TUBE (h_i vs. Re)

m (kg/sec)	T_1	T_2	T_3	T_4	LMTD	U_i	Re	hi_{exp}	hi_{theo}	% diff
0.0601	26.3	41.4	69.6	66.6	33.89	640.3	4686	1003	885.5	-13.22
0.0890	26.4	39.0	69.0	65.2	34.21	793.2	67901	1436	1349	-6.44
0.0966	26.3	38.0	70.1	65.7	35.63	823.1	7294	1537	1458	-5.45
0.1275	26.2	36.5	69.3	64.6	35.53	918.9	9479	1909	1889	-1.05
0.1550	26.6	35.7	70.2	65.0	36.42	977.3	11478	2179	1971	-10.56
0.1940	26.6	33.9	67.9	62.7	35.04	1018	14113	2391	2345	-1.95
0.2379	26.6	33.5	70.9	64.8	37.80	1100	17238	2902	2757	-5.25
0.2665	26.4	32.7	69.9	63.6	37.20	1149	19118	3268	3009	-8.61
0.3020	26.4	32.0	67.8	61.8	35.60	1176	21512	3495	3318	-5.33
0.3390	26.5	32.0	71.0	64.2	38.35	1220	24172	3918	3640	-7.62

A.3.2 HEAT TRANSFER COEFFICIENT vs. Re FOR 8mm INSERTS

m (kg/sec)	T_1	T_2	T_3	T_4	LMTD	U_i	Re	h_a	h_o	$R_1=h_a/h_o$
0.0601	26.2	45.4	74.9	70.9	36.58	772.0	4856	1368	906.5	1.51
0.089	26.2	41.9	75.3	70.4	38.55	892.0	6965	1796	1367	1.31
0.0966	26.2	38.8	69.7	65.2	34.79	883.9	7343	1764	1463	1.21
0.1275	26.2	37.7	71.4	66.0	36.67	1008	9590	2340	1898	1.23
0.1550	26.1	36.1	69.3	63.9	35.45	1072	11467	2717	1970	1.38
0.1940	26.1	35.4	73.9	67.3	39.83	1138	14254	3183	2353	1.35
0.2379	26.1	33.4	69.2	63.1	36.40	1175	17135	3489	2752	1.27
0.2665	26.2	33.2	70.5	63.9	37.50	1230	19176	4017	3012	1.33
0.3020	26.2	33.3	73.8	66.4	40.35	1297	21752	4845	3330	1.45
0.3390	26.2	32.2	70.7	63.8	38.05	1294	24147	4802	3639	1.32

A.3.3 HEAT TRANSFER COEFFICIENT vs. Re FOR 10mm INSERTS

m (kg/sec)	T_1	T_2	T_3	T_4	LMTD	U_i	Re	h_a	h_o	$R_1=h_a/h_o$
0.0601	25.8	45.5	72.7	68.8	34.50	819.2	4843	1524	904.9	1.68
0.0890	26.0	43.1	74.2	69.3	36.86	974.9	7030	2167	1373	1.58
0.0966	26.0	39.5	69.8	65.0	34.47	953.9	7378	2066	1466	1.41
0.1275	26.0	38.7	72.2	66.6	36.94	1071	9664	2710	1905	1.42
0.1550	26.0	38.3	75.9	69.0	40.24	1184	11703	3571	1983	1.80
0.1940	26.1	35.5	71.9	65.5	37.88	1185	14268	3578	2353	1.52
0.2379	26.0	33.5	69.1	62.7	36.15	1228	17135	4004	2752	1.46
0.2665	26.0	34.0	73.4	66.1	39.75	1305	19291	4945	3018	1.64
0.3020	26.0	34.1	77.5	69.3	43.35	1358	21883	5820	3337	1.74
0.3390	26.0	33.2	76.6	68.5	42.95	1361	24343	5877	3649	1.61

A. 3.4 HEAT TRANSFER COEFFICIENT vs. Re FOR 8mm INSERTS WITH β = 30cm

m (kg/sec)	T_1	T_2	T_3	T_4	LMTD	U_i	Re	h_a	h_o	$R_1=h_a/h_o$
0.0601	26.9	38.8	58.7	55.8	24.12	785.9	4599	1412	873.9	1.62
0.0890	26.9	39.1	63.3	59.5	28.19	946.8	6830	2033	1353	1.50
0.0966	26.9	39.6	67.8	63.1	32.03	985.4	7448	2220	1473	1.51
0.1275	26.9	38.0	70.5	65.0	35.23	1041	9682	2526	1906	1.33
0.1550	26.8	36.1	68.4	62.9	34.16	1083	11546	2788	1975	1.41
0.1940	26.9	35.3	68.9	62.9	34.79	1181	14352	3543	2358	1.50
0.2379	26.8	34.5	70.2	63.9	36.40	1227	17445	3987	2768	1.44
0.2665	26.9	33.5	67.8	61.9	34.65	1222	19369	3941	3022	1.30
0.3020	26.9	33.4	69.3	62.8	35.90	1309	21927	5003	3339	1.50
0.3390	26.8	32.6	69.3	62.6	36.25	1316	24392	5117	3652	1.40

A. 3.5 HEAT TRANSFER COEFFICIENT vs. Re FOR 8mm INSERTS WITH β = 20cm

m (kg/sec)	T_1	T_2	T_3	T_4	LMTD	U_i	Re	h_a	h_o	$R_1=h_a/h_o$
0.0601	32.9	48.3	71.8	67.8	28.83	880.8	5270	1751	948.4	1.85
0.0890	32.6	47.1	74.4	69.8	32.00	1001	7709	2299	1426	1.61
0.0966	27.3	40.9	70.9	66.0	34.17	976	7567	2172	1484	1.46
0.1275	29.2	39.4	68.0	63.2	31.22	1052	10024	2586	1716	1.51
0.1550	29.8	39.3	69.7	64.5	32.50	1119	12242	3035	2009	1.51
0.1940	30.2	38.8	71.9	65.9	34.38	1209	15309	3806	2403	1.58
0.2379	30.5	38.2	73.9	67.2	36.20	1272	18721	4505	2827	1.59
0.2665	30.6	37.4	72.1	65.8	34.95	1271	20837	4490	3091	1.45
0.3020	30.4	36.4	69.1	63.1	32.70	1326	23350	5270	3405	1.55
0.3390	30.2	35.8	71.3	64.5	34.90	1353	26014	5720	3727	1.53

A. 3.6 HEAT TRANSFER COEFFICIENT vs. Re FOR 8mm INSERTS WITH β = 10cm

m (kg/sec)	T_1	T_2	T_3	T_4	LMTD	U_i	Re	h_a	h_o	$R_1=h_a/h_o$
0.0601	31.7	44.7	60.7	57.8	20.64	958.4	5064	2087	929.3	2.25
0.0890	32.1	43.5	64	60.3	24.15	1054	7448	2604	1408	1.85
0.0966	32.2	43.8	67	62.7	26.68	1081	8111	2775	1528	1.82
0.1275	32.7	42.9	69.5	64.5	29.12	1151	10669	3286	1742	1.89
0.1550	32.8	42.2	69.5	64.5	29.45	1205	12903	3764	2035	1.85
0.1940	33.1	40.4	68.2	62.9	28.79	1251	15940	4251	2428	1.75
0.2379	33.1	39.9	69.7	63.9	30.30	1328	19461	5308	2855	1.86
0.2665	30.2	37.2	71.9	65.2	34.85	1333	20721	5380	3086	1.74
0.3020	27.9	34.9	73.2	65.8	38.10	1364	22474	5928	3366	1.76
0.3390	27.4	33.1	68.7	61.5	34.84	1407	24662	6825	3665	1.86

A. 3.7 HEAT TRANSFER COEFFICIENT vs. Re FOR 10mm INSERTS WITH β = 30cm

m (kg/sec)	T_1	T_2	T_3	T_4	LMTD	U_i	Re	h_a	h_o	$R_1=h_a/h_o$
0.0601	27.0	43.1	64.4	61.0	27.16	877.7	4790	1739	898.7	1.94
0.0890	27.0	41.8	66.9	62.8	30.13	1016	7010	2381	1371	1.74
0.0966	27.0	40.5	68.7	63.0	31.94	1127	7518	3095	1479	2.09
0.1275	27.0	38.8	69.3	63.9	33.60	1115	9765	3005	1913	1.57
0.1550	27.0	38.2	71.5	65.5	35.84	1184	11804	3564	1988	1.79
0.1940	26.9	36.8	73.5	66.5	38.13	1264	14563	4403	2369	1.86
0.2379	26.9	34.5	69.0	62.7	35.15	1262	17462	4384	2769	1.58
0.2665	26.9	33.8	68.1	61.8	34.60	1293	19426	4783	3025	1.58
0.3020	26.9	33.5	71.2	63.9	37.35	1342	21949	5531	3340	1.66
0.3390	26.9	33.0	70.1	63.1	36.65	1365	24515	5939	3658	1.62

A. 3.8 HEAT TRANSFER COEFFICIENT vs. Re FOR 10mm INSERTS WITH β = 20cm

m (kg/sec)	T_1	T_2	T_3	T_4	LMTD	U_i	Re	h_a	h_o	$R_1=h_a/h_o$
0.0601	33.8	52.1	73.4	69.5	27.88	976.0	5470	2173	963.2	2.26
0.0890	34.2	49.9	76.1	71.4	31.38	1074	7988	2725	1441	1.89
0.0966	34.3	49.4	78.2	72.7	33.37	1116	8642	3012	1558	1.93
0.1275	34.5	46.5	76.4	70.8	33.00	1166	11162	3407	1756	1.94
0.1550	34.5	44.6	73.6	68.1	31.24	1234	13360	4066	2048	1.99
0.1940	34.4	42.2	70.1	64.9	29.18	1263	16373	4403	2442	1.80
0.2379	34.4	40.7	68.3	63.1	28.15	1303	19822	4930	2867	1.72
0.2665	34.4	40.6	67.4	62.8	27.60	1324	22186	5233	3139	1.67
0.3020	34.4	39.7	68.0	62.6	28.25	1368	24945	6004	3463	1.73
0.3390	34.3	39.3	70.7	64.3	30.69	1410	27879	6910	3795	1.82

A. 3.9 HEAT TRANSFER COEFFICIENT vs. Re FOR 10mm INSERTS WITH β = 10cm

m (kg/sec)	T_1	T_2	T_3	T_4	LMTD	U_i	Re	h_a	h_o	$R_1=h_a/h_o$
0.0601	31.3	45.0	61.2	58.1	21.06	997.2	5059	2281	928.8	2.46
0.0890	31.5	44.2	64.2	60.3	24.13	1143	7454	3223	1409	2.29
0.0966	31.8	43.4	66.8	62.2	26.75	1118	8056	3027	1524	1.99
0.1275	31.9	42.8	69.0	63.9	29.00	1206	10587	3778	1739	2.17
0.1550	32.1	42.0	71.1	65.3	31.10	1261	12803	4368	2031	2.15
0.1940	32.3	40.6	70.7	64.8	31.28	1295	15856	4806	2425	1.98
0.2379	32.5	39.1	68.0	62.4	29.40	1325	19221	5259	2847	1.85
0.2665	32.7	38.8	68.5	62.6	29.80	1366	21512	5954	3117	1.91
0.3020	32.8	38.4	69.2	63.0	30.50	1395	24312	6560	3442	1.91
0.3390	32.6	37.9	71.8	64.7	33.00	1424	27119	7252	3770	1.92